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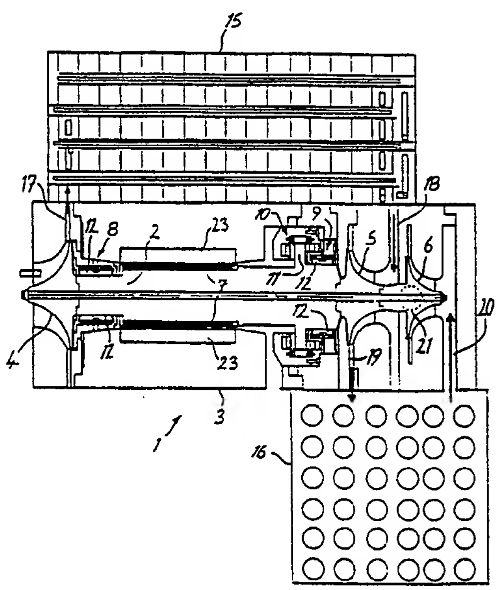

(57) Abstract

A compressor (1) comprises a rotatable shaft (2) driven by an electric motor (7, 23) having a rotor (7) mounted on the shaft (2). The shaft (2) carries at least two impeller rotor stages (4, 5, 6) arranged with the motor positioned intermediately between the impeller rotor stages. An intercooler (15) is connected intermediate the impeller rotor stages (4, 5, 6), and the shaft (2) is supported by bearing means (8, 9, 10) comprising at least one tilting pad journal bearing arranged to be self generating and air or gas lubricated, and having bearing pads provided with a ceramics bearing surface. The compressor (1) is extremely efficient and particularly suitable for use in "dry air" applications where it is important to avoid contamination of the working gas, because the bearings are not lubricated with oil.

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Compressor

The present invention relates to a compressor.

When processing food, pharmaceutical and other sensitive material it is desirable to have a supply of compressed air or other working gas which is absolutely clean or "dry", that is to say completely free of oil or other bearing lubricating material.

In the past, there have been many attempts to produce oil-free compressors, but constructions such as dry screw compressors are expensive, inefficient, use large amounts of power and are cumbersome.

The overall market for air compressors comprises a number of performance bands with each performance band encompassing in combination a range of delivery pressures and a range of mass flows.

A delivery pressure of around 8.5 bara combined with a mass flow of 0.27 kg per second is within one of the market bands for a dry air compressor. Delivery pressures can be met without difficult at the present time, but the mass flow from a conventional turbo compressor of this sort is far greater than the mass flow which is required.

In addition, turbo compressors mounted on known oil lubricated, roller or ball journal bearings would be prohibitively inefficient at the high shaft rotational speeds (typically 50,000 to 100,000 rpm) required for the desired performance. Known turbo compressors operating in this band would therefore be extremely expensive, large and inefficient.

According to the invention, there is provided a compressor comprising:

(a) a rotatable shaft;

(b) drive means arranged to rotate the shaft, the drive means comprising an electric motor having a rotor mounted on the shaft;

(c) at least two impeller rotor stages mounted on longitudinally spaced portions of the shaft such that the electric motor is positioned between said spaced portions;

(d) intercooler means provided intermediate the impeller rotor stages; and

(e) bearing means provided for the shaft, the bearing means comprising at least one tilting pad journal bearing arranged to be self generating and air or gas lubricated, and having bearing pads provided with a ceramics bearing surface.

The bearing pads may comprise homogenous pads of ceramics material.

It is preferred that the shaft is provided with hardened or ceramics surface portions against which the ceramics bearing surface of the respective tilting pads of the bearing means is arranged to act.

Advantageously the bearing means comprises at least two journal bearings, each being tilting pad journal bearings arranged to be air or gas lubricated and having bearing pads provided with respective ceramics bearing surfaces. Desirably, the journal bearings are provided to support spaced portions of the shaft advantageously adjacent opposed ends of the electric motor. It is preferred that at least one journal bearing is provided intermediately between a respective end of the motor and a respective impeller rotor stage.

It is further preferred that the shaft is supported by at least one thrust bearing which is also preferably a self generating air or gas lubricated bearing. Such a thrust bearing preferably comprises tilting pads acting against hardened, or ceramics surfaced portions of the shaft (or a thrust collar provided thereon). It is preferred that the thrust bearing is arranged to counteract axial shaft thrust acting in mutually opposed axial directions.

Advantageously, the impeller rotor stages are overhung at opposed ends of the shaft. It is preferred that each impeller rotor stage comprises a respective compressor impeller, with intercooler means being communicatively connected intermediate the impeller rotor stages.

Desirably, three impeller rotors are provided such that the compressor comprises three compression stages. It is preferred that respective intercooler means is provided intermediately between successive compressor stages. This improves the efficiency of the compressor. Advantageously, the flow of working gas into each respective impeller rotor is axial, and preferably in the direction of the electric motor.

It is accordingly preferred that at least two of the impeller stages are arranged in reverse formation relative to one another such that the respective flows into the respective impeller stages are in opposed directions, preferably towards one another. This has the advantage that the axial thrust load applied to the shaft by the respective impeller stages tend to cancel each other out, thereby reducing the axial thrust taken up by the thrust bearing.

It is preferred that seal means, preferably comprising respective labyrinth seals, are provided for the shaft, arranged to inhibit access of the working gas from the impeller rotor stages to the motor and bearing means.

Advantageously, the electric motor comprises an electromagnetic or permanent magnet electric motor, preferably arranged to rotate the shaft at over 50,000 r.p.m. and more preferably at over 70,000 r.p.m. Desirably the electric motor is a direct current motor, preferably controlled by a variable frequency source.

The invention will now be further described in a specific embodiment by way of example only, and with reference to the accompanying drawings, in which:

Figure 1 is a schematic representation of a compressor according to the invention; and

Figure 2 is an enlarged detail of a part of the compressor of Figure 1.

Referring to the drawings, there is shown a compressor generally designated 1. The compressor 1 comprises an axial rotatable shaft 2 mounted in a housing 3, and having machined aluminium impeller rotors 4,5,6 mounted thereon.

Intake, first stage, rotor 4 is overhung at one end of the shaft, whereas second and third stage rotors 5 and 6 respectively are overhung at the opposed end. Intermediately between impeller rotors 4 and 5 there is positioned a brushless D.C. motor having a rotor 7 comprising permanent magnets mounted on the shaft 2 and a stator 23 mounted in the housing. A solid state thyristor based inverter/controller (not shown) is used to generate a variable but high frequency current from a standard 415V/50Hz electrical supply. The high frequency current drives the motor (and therefore directly drives the shaft 2 without the need for intermediate gearing) at the required high operational speed which is typically of the order of 50,000 to 100,000 r.p.m. Because no gearing is required to couple shaft 2 to the drive, power losses are minimised.

The shaft 2 is supported in housing 3 on journal bearings 8,9 provided at either end of the electric motor, adjacent impeller rotors 4 and 5 respectively. A thrust bearing 10 is also mounted in the housing to act on thrust collar 11 provided on the shaft. Journal bearing 8,9 comprise tilting pad journal bearings which are self generating and air lubricated. The tilting pads 12 of each journal bearing 8,9 are supported on flexible pivots 24, and provided with ceramics bearing surfaces 13 which are arranged to act on immediately

adjacent bearing surface portions of the shaft. The bearing surface portions of the shaft are coated with hardened deposit to increase wear resistance.

It is an important feature of the design that frictional losses in the bearings are minimised to maximise the efficiency of the compressor. Typically, where fluid lubricated journal bearings (such as oil lubricated bearings) or ball or roller journal bearings are used in high speed rotating machinery frictional losses in the bearings amount to between 5% and 10% of the driving power. The provision of tilting pad self generating air (or gas) bearings cuts frictional losses to approximately 0.5% of driving power. However due to the fact that the shaft rotation speed is extremely high (e.g. 80,000 r.p.m. for a compression from 1 bara to 8.5 bara at a mass flow of 0.27 kg/s for air) the temperature generated at the bearings is extremely high, which can cause problems with bearing/shaft material expansion due to the necessarily small bearing shaft clearances required for the operation of air or gas lubricate tilting pad self generating journal bearings (typically 0.003" diametral clearance for journal bearings). This problem is overcome by utilising ceramics materials for the bearing surfaces of tilting pads 12; the provision of a hardened deposit surface covering for the bearing portions of the shaft 2 also assists in overcoming this problem.

Thrust bearing 10 is also provided with tilting pad thrust members 10a,10b provided with ceramics bearing surfaces. Pads 10a are arranged to take up normal thrust loading transferred from shaft 2 by thrust collar 11 during normal running of the compressor. Pads 10b act on the opposite side of collar 11 and act to take up reverse thrust loading during motor and shaft "run up" to normal operational speed.

To increase efficiency, an intercooler 15 is provided intermediately between first stage impeller 4 and second stage impeller 5. A second intercooler 16 is provided intermediately between second stage impeller 5 and final (third) stage impeller 6. It is an important feature of the compressor that the flow of working gas into the first stage impeller 4 is in an opposed direction to the flow of working gas into the second and third stage impellers 5,6. This has the effect of "balancing" the axial thrust acting on the shaft and reducing the usual axial thrust applied to thrust bearing 10. Bearing losses in thrust bearing 10 are thereby minimised.

In operation, the electric motor is run up to an operating speed of around 80,000 r.p.m. Working gas is then drawn axially into the first impeller stage 4 and forced

out through duct 17 into intercooler 15. The working gas leaves intercooler 15 entering duct 18 and subsequently passing axially into second impeller stage 5. The working fluid leaves impeller 5 radially passing via duct 19 into second intercooler 16. Intercoolers 15 and 6 are substantially identical, except that intercooler 16 is arranged with its longitudinal dimension at 90° to the longitudinal dimension of intercooler 15 (i.e. the longitudinal dimension of intercooler 16 is out of the page in Figure 1).

Working gas leaves intercooler 16 via duct 20 and is directed to enter the third (and final) impeller stage 6 axially. The working gas leaves the final impeller stage 6 radially via outlet duct 21 (the outlet flow through duct 21 is out of the page in Figure 1).

Due to the combination of the high speed directly driven rotatable shaft, together with the minimisation of bearing losses and he split stage intercooled arrangement of the impeller rotors, an extremely efficient compressor is provided according to the invention. The compressor enables a compact turbomachine to be used in applications previously served mainly be screw feed type compressors since, unusually for a turbo compressor high delivery pressures (8.5 bara typically) are achievable with relatively low mass flows (0.27 kg/s typically for air).

Claims:

1. A compressor comprising:

(a) a rotatable shaft;

(b) drive means arranged to rotate the shaft, the drive means comprising an electric motor having a rotor mounted on the shaft;

(c) at least two impeller rotor stages mounted on longitudinally spaced portions of the shaft such that the electric motor is positioned between said spaced portions;

(d) intercooler means provided intermediate the impeller rotor stages; and

(e) bearing means provided for the shaft, the bearing means comprising at least one tilting pad journal bearing arranged to be self generating and air or gas lubricated, and having bearing pads provided with a ceramics bearing surface.

2. A compressor according to claim 1, wherein the shaft is provided with hardened or ceramics surface portions against which the ceramics bearing surface of the respective tilting pads of the bearing means is arranged to act.

3. A compressor according to claim 1 or claim 2, wherein two journal bearings are provided to support spaced portions of the shaft, both journal bearings being tilting pad journal bearings arranged to be air or gas lubricated and having bearing pads provided with respective ceramics bearing surfaces.

4. A compressor according to any preceding claim, wherein at least one journal bearing is provided intermediately between a respective end of the motor and a respective impeller rotor stage.

5. A compressor according to any preceding claim, wherein the shaft is supported by at least one thrust bearing.

6. A compressor according to claim 5, wherein the thrust bearing is arranged to counteract axial shaft thrust acting in mutually opposed axial directions.

7. A compressor according to claim 5 or claim 6, wherein the thrust bearing is a self acting and air or gas lubricated, tilting pad bearing, having bearing pads provided with a ceramics bearing surface.

8. A compressor according to any preceding claim, wherein the impeller rotor stages are overhung at opposed ends of the shaft.

9. A compressor according to any preceding claim comprising more than two impeller rotor stages.

10. A compressor according to claim 9, wherein a respective intercooler is connected intermediately between successive impeller rotor stages.

11. A compressor according to any preceding claim, wherein at least two of the impeller stages are arranged in reverse formation relative to one another such that the respective working gas flows into the respective impeller stages are in opposed directions.

12. A compressor according to any preceding claim, wherein seal means is provided for the shaft, arranged to inhibit access of the working gas from the impeller rotor stages to the motor and bearing means.

13. A compressor according to any preceding claim, wherein the electric motor is arranged to rotate the shaft at a speed of over 50,000 r.p.m.

14. A compressor according to any preceding claim, wherein the electric motor is arranged to drive the shaft directly without being coupled to intervening gearing.

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FIGURE 1

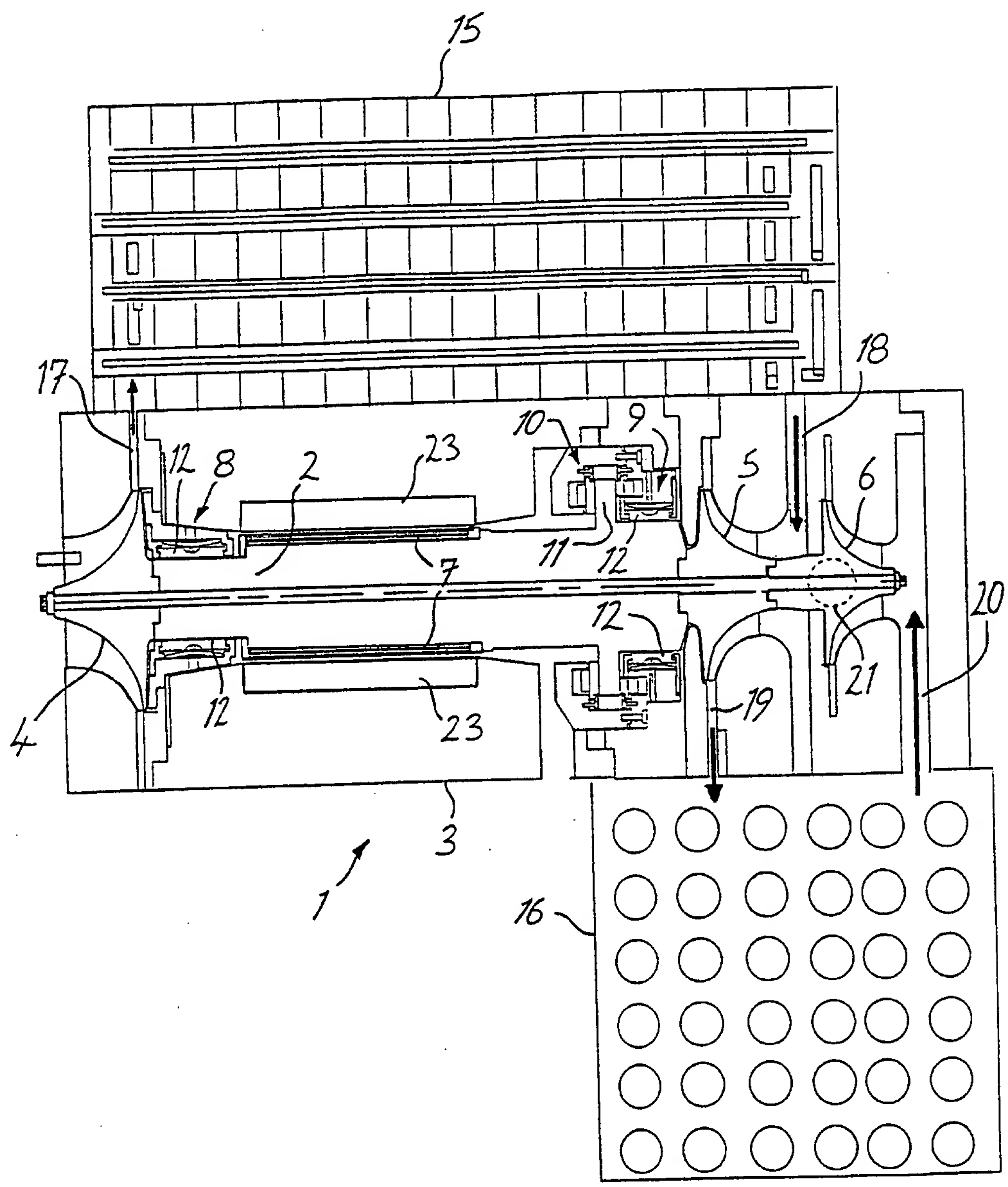

FIGURE 2